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Energetic and exergetic analyses of a dual-fuel diesel engine

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ABSTRACT

The aim of this work is to investigate theoretically and experimentally the performance characteristics of a commercial diesel engine when operating in dual form: natural gas and diesel. The experimental facility (thermal system) is composed of a diesel engine coupled to an electronic generator with measuring sensor for temperature and pressure, air, natural gas and diesel flow meters, gas transducers, gas analyzer and power absorption system, constituted by an electric charge bank and its controlling system. For energetic and exergetic analysis of such dual engine, a mathematical model based on the thermodynamics concepts was developed. Numerical and experimental results concerning the effect of air conditions, the type and quantity of fuel used and the exhaustion gases over the engine performance and environmental impact are presented and analyzed. In this work, the diesel engine operated with powers ranging from 10 to 150 kW and replacement rates from 60% to 85%.

Contents

1. Introduction

For centuries energy saving has been a basic requirement for economic development of the countries in the world, as a result of the growing energy consumption for the population and industries. Special attention is given to energy related to use of oil and gas. The increased use of fossil fuels causes serious environmental problems. In this context, the world has great interest concerned

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over the environmental impact and/or exhaust emissions of fossil fuel combustion mainly those related to internal combustion engine. Diesel engines have been used for decades in different situations such as, industrial power, cogeneration, power systems, locomotives, truck, stationary and agricultural power stations. The complex and transient nature of the diesel engine combustion process has been the focus of many researches. In the last years there studies are related mainly to exhaust emissions and energy consumption. In this sense, natural gas appear as an attractive energy source to be used as a fuel in combustion process for power production in internal combustion engines due to the physical and chemical properties and to be available in great quantities in many location of the world, thus resulting in efficient combustion and substantial reduction of emissions [1]. Due to this important characteristic natural gas has been used as alternative fuel in diesel engine so called as dual-fuel engine. Dual fuel engines presents different good attributes: (a) operate with more than one fuel source, (b) it has gaining popularity because they reduce the amount of diesel fuel used, (c) they reduce pollutant emissions, thus improvement of air quality, (d) does not require necessary modifications in diesel engine to operate in dual mode, (e) full original power capacity (in dual or pure diesel modes), (f) diesel pilot fuel provides lubrification to valves and rings, when combined with natural gas. Because of these advantages, dual fuel engines are becoming popular in many parts of the world. In dual-fuel engine a mixture of air, diesel and natural gas is compressed and then fired by ignition at the end of compression phase. According to Mansour et al. [2] the advantage of this type of engine resides in the fact that it uses the difference in flammability of two fuels. In addition we can cite the economical and environmental potential benefits of using natural gas in diesel engine. The use of natural gas as fuel in CI engine presents many advantages such as: less pollutant combustion (clean combustion) on various proportions of the gas/diesel mixture, fuel flexibility and more fuel cost saving.

The performance of dual-fuel engine (gas–diesel) has been investigated by many researchers with promising results [3–18] but few works have been carried out on energy and exergy analysis [19–21]. For diesel engine operating in pure diesel mode different works are reported in the literature [22–24].

Sahoo et al. [18] reports a review related to research carried out by various scientists about the effect of engine operating and design parameters (load, speed, compression ratio, pilot fuel injection timing, pilot fuel mass inducted, intake manifold conditions) and type of gaseous fuel on the performance of dual fuel (gas–diesel) engines. Comment about performance, combustion and emission characteristics of different dual-fuel engines which use natural gas, biogas, produced gas, methane, liquefied petroleum gas, propane, etc, as gaseous fuel are presented. The authors reveal that thermal efficiency of dual-fuel engines improves with either increased engine speed, or with advanced injection timings, or with increased amount of pilot fuel.

Due to the fast depletion of fossil fuels and increase in demand of energy with less pollutant emissions, it is required to explore different resources of energy such as liquefied petroleum gas, biogas, producer gas, biodiesel, etc. In this sense, Lata and Misra [12], Lata et al. [16], Lata and Misra [15] and Lata et al. [17] present theoretical and experimental studies related to use of hydrogen and liquefied petroleum gas as secondary fuels in dual fuel diesel engines. These papers contain reported information about the ignition delay, combustion, efficiency and emissions pollutant. Papagiannakis et al. [13] and Papagiannakis et al. [14] present studies about the effects of the engine parameters (total air–fuel ratio and air inlet temperature) on performance and emissions of dual diesel engine (natural gas and diesel). According to the authors, the increase in air temperature intake could be a promising solution for improving engine efficiency and reducing CO emissions. The use of natural gas as a supplement for liquid diesel fuel permits to control both NO and soot emissions on existing direct injection diesel engine, requiring only slight modifications of the engine structure.

Carlucci et al. [11] reports an experimental investigation and combustion analysis of a direct injections dual-fuel diesel–natural gas engine. In this research the effect of compressed natural gas (methane) and diesel fuel injection pressure and quantity of fuel injected during the pilot injection were analyzed on the combustion development and engine performance (emissions and fuel consumption). It was verified that an analysis of the rate of heat release is not sufficient to explain the effect of each of the injection parameters on the pollutant emissions.

The literature contains many papers and textbooks for second law analysis of thermodynamic systems [19,25–28]. Historically speaking, thermodynamics has influenced ideas about energy and exergy conversion.

Szargut [28] defines exergy as the amount of work obtainable when some matter is brought to a state of thermodynamic equilibrium with the common components of the natural surroundings by means of reversible process, involving interaction only with this component of nature. According to Kotas [27], the exergy is the energy that can be completely converted into mechanical energy. It can be established as the most appropriate standard for evaluating the variation in the quality of energy in the analysis of thermal systems. When exergetic analysis is carried out, it is possible to identify the points where losses occur, i.e., where the destruction of exergy exists. This destruction of exergy is a function of the irreversibility of the process or degradation of the quality of energy resources, then, exergy analysis can be used to indicate possible ways of improving of thermal and chemical processes and therefore, what areas should receive special technical. However, exergy analysis cannot state whether or not the possible improvement is practicable [28] Sorathia and Yadav [21] reports an energy and exergy analyses applied to internal combustion engine operating in dual mode (diesel and biogas). According to the authors the energetic and exergetic performance has presented similar trends. The exhaust energy loss by using diesel oil is more than the biogas mode and the diesel–biogas dual fuel mode produced lower energy efficiency. Besides, using the exergy as a measure quality, the diesel fuel is a greater quality fuel than biogas, because calorific value of diesel is greater than that of the biogas. Kanoglu [22] presents an investigation directed to exergetic performance of a turbocharged stationary diesel engine. In this study, the exergy destruction and exergy efficiency are determined as a function of inlet air temperature and pressure. The exergy efficiency of the engine was found to be 40.5%. Caliskan et al. [23] presents an excellent review on energy and exergy analysis of Otto and diesel engines from 1963 to 2008. The test engines had different cylinders numbers, speeds and rated powers. According to the authors, the best exergetic efficiency for a turbocharged diesel engine was about 30%.

Razac and Meghalchi [29] reports that it is important that scientists and engineers continue to innovate and improve current methods on energy production and energy systems. In this sense, the authors recommend the use of exergy analysis as a new tool for measure efficiency of the studied system. According to Ameri et al. [24] and Abassi et al. [30], the use of a combined energy and exergy analysis provides better criteria for the performance assessment of thermal system. Rosen [31] reports about the benefits that exergy provides to industry, government and society, in area like energy, environment, economic development and design. The author demonstrates that exergy can help improve understanding of energy issues and thus help increases the utilization of beneficial energy resources and technologies. In the next paper, Rosen [32] presents some of the needs related to exergy and its use in different sectors (industry, public and government).

From this review we can see that the use of the scientific concept of energy to analyze performance of process is destined

to lead to poor decisions. In this order, the exergy method is the best technique of thermodynamic analyses and its use is constantly growing, in the recent years. In this sense, the aim of this work is to describe a theoretical and experimental study to analyze the energetic and exergetic performance of a dual-fuel diesel engine when liquid diesel is partially substituted by natural gas under ambient intake temperature.

2. Experimental methodology

2.1. Description of natural gas and diesel fuel composition used in this experiment.

The chemical composition of the natural gas and diesel fuel used in the experiment are presented in Table 1. These values are representatives of typical commercial fuels supplied in Campina Grande City, Paraiba State, Brazil.

Table 1
Basic composition of diesel and natural gas used.

Fuel (source)	Chemical composition (in volume)						
Diesel—[33]	$C_{12}H_{26}$ 98.53%				S 1.47%		
Natural gas—[34]	CH_4 89.42%	C_2H_6 7.24%	C_3H_8 0.16%	C_4H_{10} 0.18%	N_2 1.27%	CO_2 1.66%	O_2 0.08%

Fig. 1. Electro-mechanical system.

2.2. Experimental apparatus and procedures

The electro-mechanical system studied consists of a commercial engine (Cummins 6CTA), with mechanical power of 188 kW @ 1800 rpm, coupled to an electric generator Onan Genset of 150 kW. The unit is totally scored with air, gas and diesel flow meters, temperature and pressure sensors in several points of the system and probe for gas analysis. Experimental data of mass flow rates, temperature and pressure of air, gas and diesel, and exhaust emissions are collected in real time through data acquisition system. Fig. 1 presents the electromechanical system. Details about the experimental apparatus and procedure has been described by Costa et al. [35] and Costa [36]. Table 2 shows a sample of the collected data in the experiments.

3. Mathematical modeling

Before the exergy analysis, mass and energy balances on the engine are required to determine the mass flow rate and energy transfer rate at the control surface.

To describe the theoretical model, consider Fig. 2, which shows an engine schematic form. In the analysis, it is considered that the fuel enters in the engine, with mass flow rate $\dot{m}_c$ and it is mixed with a quantity of air $\dot{m}_a$. Both the air and fuel have kinetic and potential energy variations negligible. The fuel enters the engine at temperature T_c and pressure P_c, while the air enters at temperature T_a and pressure P_a. The mixture burns completely and the combustion products leave the engine at temperature T_p and pressure P_p with mass flow $\dot{m}_f$. The engine develops a power output and transfers a quantity of heat to the environment $\dot{Q}$.

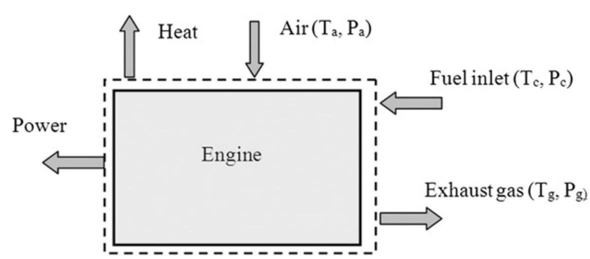

Fig. 2. Schematic of the fuel diesel engine.

Table 2
Experimental data.

Power (kW)	Temp. water input (°C)	Temp. water output (°C)	Temp. lubricating oil (°C)	Air pressure in the intake manifold (bar)	Ambient pressure (bar)	Motor speed (rpm)	Mass flow rate of diesel (kg/h)	Mass flow rate of air (kg/h)	Mass flow of gas (kg/h)	Ambient Temp. (°C)	Temp. fuel oil (°C)
140.3	73.48	87.23	131.64	2.52	0.98	1772.81	32.43	1491.85	14.34	33.0	37.0
132.5	74.67	86.23	133.15	2.43	0.98	1758.22	31.17	1440.28	16.59	33.0	39.0
122.9	74.94	86.01	132.05	2.32	0.98	1755.34	27.59	1369.33	16.15	35.0	42.0
111.1	74.52	86.00	131.43	2.22	0.98	1755.31	26.97	1311.31	15.85	34.0	42.0
103.3	73.92	86.00	130.23	2.11	0.98	1754.92	25.44	1265.14	14.97	35.0	42.0
93.4	73.46	86.00	129.16	2.00	0.98	1754.84	24.16	1197.84	14.40	34.0	39.0
86.7	73.14	85.88	127.66	1.89	0.98	1752.84	22.29	1119.19	14.01	34.0	42.0
77.2	73.33	85.58	126.24	1.79	0.98	1749.87	19.73	1065.95	13.14	33.0	42.0
68.2	73.41	84.21	125.33	1.69	0.98	1745.90	17.22	1010.13	12.55	33.0	42.0
58.0	73.51	83.46	124.13	1.60	0.98	1744.43	15.32	947.19	11.81	32.0	39.0
48.1	73.89	82.65	124.00	1.52	0.98	1742.07	14.52	903.34	11.33	32.0	42.0
40.0	73.93	82.10	124.00	1.44	0.98	1733.01	11.92	852.90	10.57	30.0	42.0
29.3	74.04	82.00	123.99	1.36	0.98	1732.28	10.17	800.40	10.01	29.0	42.0
19.2	74.86	82.00	123.25	1.30	0.98	1730.36	7.91	755.93	9.42	28.0	42.0
11.7	75.04	82.00	122.20	1.24	0.98	1727.24	6.53	717.48	8.59	28.0	42.0

All the exchanges of energy between the lubrication oil and refrigeration water with the body of the engine, and air with the turbo-compressor and cooler, are contained within the control volume that involves the engine, and therefore their effects are contained in $\dot{Q}$. The internal combustion engine operates at steady-state regime.

3.1. Mass balance

For steady state situation, all properties are unchanged in time. In this case, for a control volume presented in Fig. 2, (dashed line), the total mass flow rate entering in the control volume (air and fuel) was assumed to equal the total mass flow rate leaving the control volume (exhaust gases). Then we can write:

$$\sum_e \dot{m}_e = \sum_s \dot{m}_s \tag{1}$$

where $\dot{m}_e$ and $\dot{m}_s$ represent inlet and outlet mass flow rate, respectively.

3.2. Energy balance

The equation of the first law of thermodynamics is used to determine the heat transfer involved in the analysis of the engine. The energy rate balance at steady-state can be written as:

$$\dot{Q}_{vc} + \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gZ_e \right) = \sum \dot{m}_s \left(h_s + \frac{V_s^2}{2} + gZ_s \right) + \dot{W}_{vc} \tag{2}$$

where $\dot{Q}$ is the heat transfer rate (kW), $\dot{m}$ is the mass flow rate (kg/h), h is the specific enthalpy (kJ/kg), $V^2/2$ is the specific kinetic energy (kJ/kg), gZ is the potential energy (kJ/kg) and $\dot{W}$ is the useful power developed by the engine (kW).

The Eq. (2) allows an assessment of the heat transfer rate that is lost by the engine to the environment. Kinetic and potential energy effects are small and therefore are neglected. So, the Eq. (2) can be written as follows:

$$\dot{Q} + \sum \dot{m}_e h_e = \sum \dot{m}_s h_s + \dot{W} \tag{3}$$

In dual-fuel engines, diesel, natural gas and air burns completely given as results combustion products. In terms of diesel, natural gas, air and combustion products composition on dry basis, the Eq. (3) can be written as:

$$\begin{aligned}
&\dot{Q} + \dot{n}_d (y_{C_{12}H_{26}} M_{C_{12}H_{26}} h_{C_{12}H_{26}} + y_s M_s h_s)_d + \dot{n}_g (y_{CH_4} M_{CH_4} h_{CH_4} \\
&\quad + y_{C_2H_6} M_{C_2H_6} h_{C_2H_6} + y_{C_3H_8} M_{C_3H_8} h_{C_3H_8} + y_{C_4H_{10}} M_{C_4H_{10}} h_{C_4H_{10}} \\
&\quad + y_{N_2} M_{N_2} h_{N_2} + y_{CO_2} M_{CO_2} h_{CO_2} + y_{O_2} M_{O_2} h_{O_2})_g \\
&\quad + \dot{n}_{O_2} (M_{O_2} h_{O_2} + 3.76 M_{N_2} h_{N_2} + 7.655 \omega M_{H_2O} h_{H_2O})_a \\
&\quad = \dot{n}_P (y_{CO_2} M_{CO_2} h_{CO_2} + y_{CO} M_{CO} h_{CO} + y_{N_2} M_{N_2} h_{N_2} + y_{NO} M_{NO} h_{NO} \\
&\quad + y_{NO_2} M_{NO_2} h_{NO_2} + y_{SO_2} M_{SO_2} h_{SO_2} + y_{CH_4} M_{CH_4} h_{CH_4} \\
&\quad + y_{O_2} M_{O_2} h_{O_2})_p + (\dot{n}_{H_2O} M_{H_2O} h_{H_2O})_{H_2O} + \dot{W}
\end{aligned} \tag{4}$$

where $\dot{n}_d$, $\dot{n}_g$, $\dot{n}_{O_2}$, $\dot{n}_P$, $\dot{n}_{H_2O}$ are molar flows of diesel, natural gas, oxygen, combustion products and water vapor, respectively.

In the Eq. (4) ω is the absolute humidity of air, and y_i and M_i are the molar fractions and molecular weights of each component i involved in the combustion process. The molar fractions for the diesel and natural gas are provided in Table 1. The molar fractions of combustion products on the dry basis, y_{CO_2}, y_{CO}, y_{NO}, y_{NO_2}, y_{CH_4} and y_{O_2} are measured with the gas analyzer. The other coefficients of the equation are obtained from a stoichiometric balance for each chemical element involved in the process.

In the molar basis Eq. (4) can be written as follows:

$$\begin{aligned}
&\dot{Q} + \dot{n}_d \left(\sum_{i=1}^{\hat{i}} y_i \bar{h}_i \right)_d + \dot{n}_g \left(\sum_{j=1}^{\hat{j}} y_j \bar{h}_j \right)_g \\
&\quad + \dot{n}_{O_2} [\bar{h}_{O_2} + 3.76 \bar{h}_{N_2} + 7.655\, \omega\, \bar{h}_{H_2O}] \\
&\quad = \dot{n}_p \left(\sum_{k=1}^{\hat{k}} y_k \bar{h}_k \right)_{\text{Dry products}} + \dot{n}_{H_2O}\, \bar{h}_{H_2O} + \dot{W}
\end{aligned} \tag{5}$$

The specific enthalpy and molar basis is calculated by:

$$\bar{h} = \bar{h}_f^0 + \Delta \bar{h} \tag{6}$$

where $\bar{h}_f^0$ is the enthalpy of formation on molar basis. The specific heat capacity in molar basis is given by:

$$\Delta \bar{h} = \int_{T_{ref}}^{T} \bar{c}_p dT \tag{7}$$

With $\bar{c}_p = M c_p$.

The second term in the Eq. (6) accounts for the change in enthalpy from the temperature T_{ref} to the temperature T. The molar specific heat capacity and enthalpy formation can be found in the literature [19,20,27,37–39].

The thermal efficiency (η_t) of the engine is calculated by the following expression:

$$\eta_t = \dot{W}/PCI \tag{8}$$

where PCI is the Lower Heating Value of the mixture (kJ/mol) and $\dot{W}$ is the useful power developed by the engine (kW)

The energy efficiency of the engine, η_t, can be written more appropriately as:

$$\eta_t = \frac{\dot{W}}{\dot{n}_d [\sum_{i=1}^{k} (y_i M_i PCI_i)]_d + \dot{n}_g [\sum_{j=1}^{r} (y_j M_j PCI_j)]_g} \tag{9}$$

where i and j denote the components of diesel and natural gas, respectively.

3.3. Exergy balance

In many applications, the means of work consists of a mixture of ideal gases, for example, gaseous fuels, combustion products, etc. When a hydrocarbon fuel C_aH_b or another substance is a component of a mixture of ideal gases in the standard state (temperature T_0 and pressure P_0), the hydrocarbon fuel or another substance is in the state (T_0, y_i P_0). In this case, the chemical exergy of the fuel or substance, $\bar{x}_i^{chem}$, is given by the follow equation [6,20,27]:

$$\bar{x}_i^{chem}(T_{0i}, y_i P_0) = \bar{x}_i(T_0, P_0) + \bar{R} T_0 \ln(\mu_i y_i) \tag{10}$$

In the Eq. (10) y_i is the molar fraction of component i in the mixture of hydrocarbon fuel, $\bar{R}$ is the universal constant gas and $\bar{x}_i(T_0, P_0)$ is the exergy in the standard state. In this equation, we consider the coefficient of activity $\mu_i = 1$.

In the ambient conditions (standard reference state), the thermodynamic exergy is null. Then, the total exergy for a fuel is exactly equal to the chemical exergy. In the case studied, the exergy is given by:

$$x_{\text{diesel or gas}}^{chem} = \frac{\bar{x}_{\text{diesel or gas}}^{chem}}{M_{\text{diesel or gas}}} = \left(\frac{\sum_{i=1}^{n} y_i \bar{x}_i^{chem}}{M} \right)_{\text{diesel or gas}} \tag{11}$$

In order to calculate exergy, the environment must be specified.

For the determination of the chemical exergy of the exhaust gas, we use the Eqs. (10) and (11), and the molar fraction of components obtained experimentally.

$$\dot{X}_{chem} = \dot{n}_p \sum_{i=1}^{\hat{i}} y_i \bar{x}_i + \dot{n}_{H_2O} \bar{x}_{H_2O}^{chem} \tag{12}$$

The exergetic efficiency for the engine is determined with the aid of an exergy balance. At steady state regime the rate at which

exergy enters the engine equals the rate at which exergy exits plus the rate at which exergy is destroyed within the engine. In this statement, the air used for combustion enters the environmental conditions, and consequently with a value zero of exergy, only fuel provides exergy to the engine. The exergy exit the engine accompanying heat and work, and with the combustion products. If the engine power developed is taken to be the product of the engine, and the heat transfer and gases produced in the output are seen as losses, an expression for the exergetic efficiency (ε), which measures how much exergy at the entrance of the engine is converted to the products, is of the form:

$$\varepsilon = \frac{\dot{W}_{vc}}{\dot{X}_c} \tag{13}$$

where $\dot{X}_c$ denotes the rate at which exergy enters with the fuel. This parameter is given by:

$$\dot{X}_c = \dot{n}_d\left(\sum_{i=1}^{\hat{i}} y_i\bar{x}_i\right)_d + \dot{n}_g\left(\sum_{j=1}^{\hat{j}} y_j\bar{x}_j\right)_g \tag{14}$$

To obtain the exhaust gas, heat, work and destroyed exergy we use the following equation:

$$\dot{X}_{\text{Physic}} = \dot{n}_p \sum_{k=1}^{j} y_k M_k[(h_k - h_{k0}) - T_0(s_k - s_{k0})] + \dot{n}_{H_2O}\left[(h_{H_2O} - h_{H_2O_0}) - T_0\left(s_{H_2O} - s_{H_2O_0}\right)\right] \tag{15}$$

$$\dot{X}_{\text{gas}} = X_{\text{Phisic}} + \dot{X}_{\text{chem}} \tag{16}$$

$$\dot{X}_{\text{heat}} = (1 - T_0/T_m)\dot{Q} \tag{17}$$

$$\dot{X}_{\text{work}} = \dot{W} \tag{18}$$

$$\dot{X}_{\text{destroyed}} = \dot{X}_{\text{total}} - \dot{X}_{\text{gas}} - \dot{X}_{\text{work}} - \dot{X}_{\text{heat}} \tag{19}$$

where j is the number of components of the exhaust gas.

In the Eq. (15) the entropy was obtained in the partial pressure of each component in the mixture. We considers T_m=313.15 K (temperature of the system boundary where heat is transferred to the environment). An adopted standard state T_0=27 °C and P_0=1.0 atm was used. These parameters were obtained by the average value monitored by the acquisitions system, during the experiments.

4. Results

4.1. Replacement rate and exhaust emissions.

The experiments analyzed in this study were those related to higher replacement rate for each power. These values were then used to evaluate engine performance. The average values of the replacement rate were 83.78%. The replacement rate was calculated as follows [4]:

$$\text{Replacement rate}(\%) = (\dot{m}_d - \dot{m}_{dg})/\dot{m}_d \times 100 \tag{20}$$

where $\dot{m}_d$ is the mass flow rate of the fuel when the engine operating with pure diesel (kg/h); $\dot{m}_{dg}$ is the mass flow rate of fuel when the engine operates in a dual form—gas and diesel—(kg/h). Additional information about exhaust emissions can be found in Costa et al. [35]. Fig. 3 presents the replacement rate as a function of the power.

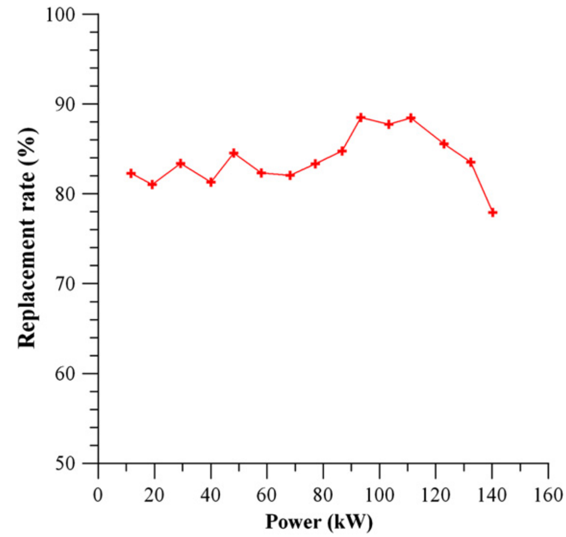

Fig. 3. Replacement rate as a function of the power.

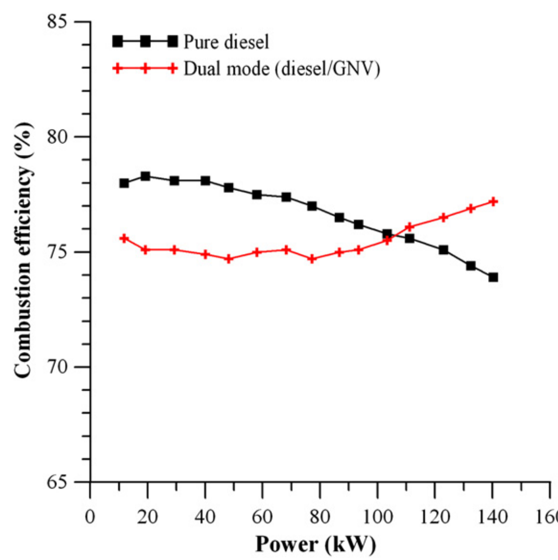

Fig. 4. Combustion efficiency as a function of the power.

4.2. Combustion efficiency as a function of applied load

The combustion efficiency (η_{comb}), expressed in %, shown in Fig. 4 is calculated by the gas analyzer as follows:

$$\eta_{\text{comb}} = 100\% - \text{DFGL} \tag{21}$$

where DFGL=20.9 × K_{1n} × Tnet/[K_2 × (20.9-%O_2)], being K_{1n}=0.515 and K_2=15.51 constants of the analyzer when pure diesel is used as combustible, and K_{1n}=0.393 and K_2=11.89 when using natural gas.

In the present work we use the constants of the natural gas for the engine operating of dual form. In Fig. 4, we can see that for low powers, the lower combustion efficiency is found (dual form), when compared to pure diesel. This efficiency increases with applied load, by presenting a value of approximately 78% in the power of 150 kW. This behavior of the combustion confirms the results of the emissions obtained during the tests and reported in the literature [35,36].

4.3. Energetic analysis

Fig. 5 shows the thermal efficiency of the engine in similar working conditions, changing only the fuel supplied to the engine. In the operation with pure diesel, we can see that the efficiency of the engine change from 30% (70 kW) to 35% (150 kW). When the engine is operated with the mixture of gas/ diesel, there is an increase in the efficiency of the engine. At higher loads, the efficiency of the engine with mixture diesel/gas is higher, reaching values of 53%. For lower powers, the efficiency decreases with the power for the condition of dual-mode operation.

Data published by Braga et al. [40] shows that the thermal efficiency of an engine of 135 kVA operating with pure diesel, presented value of 22% for loads smaller than 50 kW and it has reached a maximum value of 34% for load of 100 kW. Henham and Makkar [41] reports thermal efficiency of 28.2% when the engine operates at 2000 rpm and 40 Nm of torque. These differences are related to the form to calculate this parameter. It can be observed that in the power output exceeding 100 kW and operating in dual mode, the combustion efficiency of engine is over. This increase in efficiency occurs because the ease that this fuel has to be mixed with air, leading to better combustion.

Fig. 6 shows the waste heat by the engine to the environment per unit of time. We can see that the system running in dual mode, provides a lower loss of heat. The heat transferred to the environment is approximately 40 kW for a power of 150 kW.

4.4. Exergetic analysis

Figs. 7–10 presented shows the heat transferred to the environment, exergetic efficiency, total exergy, destroyed exergy, exergy of heat and the exergy of the combustion products, all expressed in terms of power. Data needed for calculations as molar fraction of exhaust gas, power, mass flow rate of gas and diesel have been measured.

Fig. 7 illustrates the exergetic efficiency as a function of power. It can be seen that there is an increase in the efficiency for loads exceeding 80 kW, when operating in dual mode. The calculated values show that a substantial portion of the fuel exergy is destroyed in the combustion process. We see therefore, that a

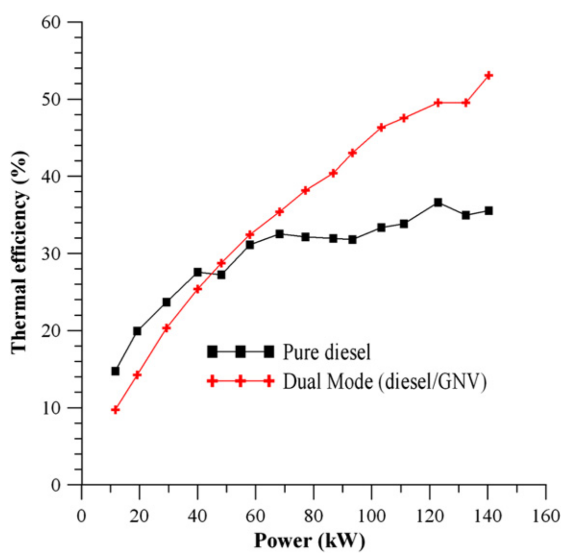

Fig. 5. Thermal efficiency as a function of the power.

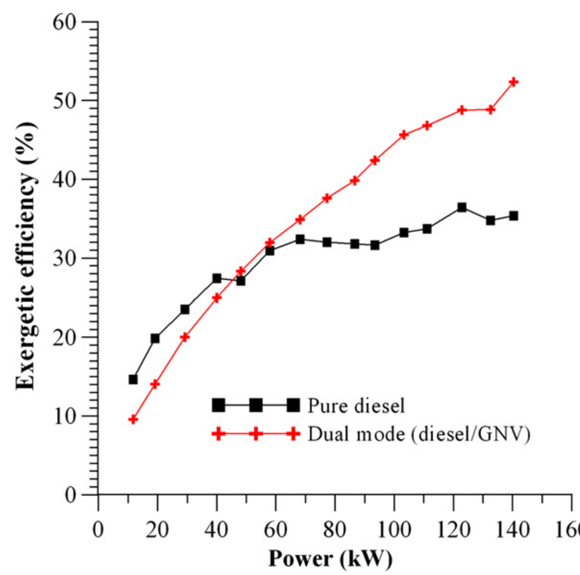

Fig. 7. Exergetic efficiency as a function of the power.

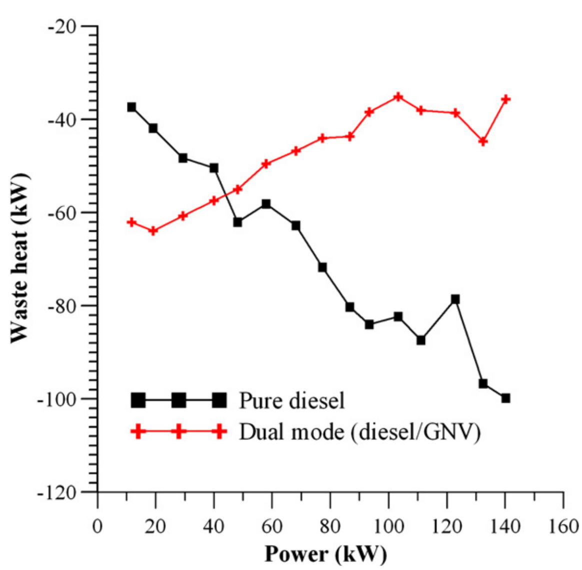

Fig. 6. Waste heat for the environment as a function of the power.

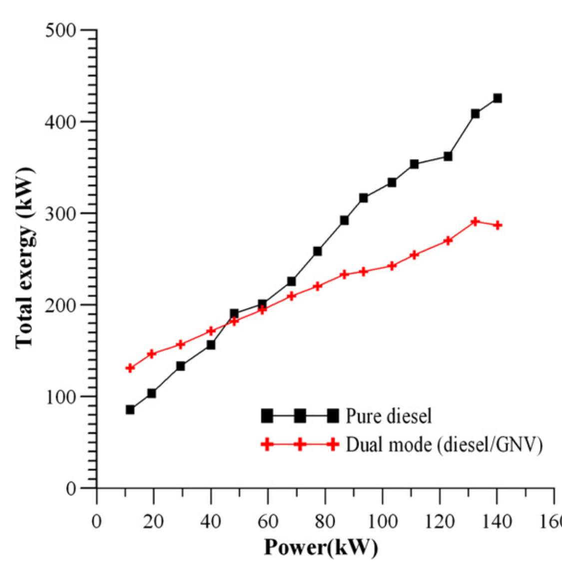

Fig. 8. Total exergy as a function of the power.

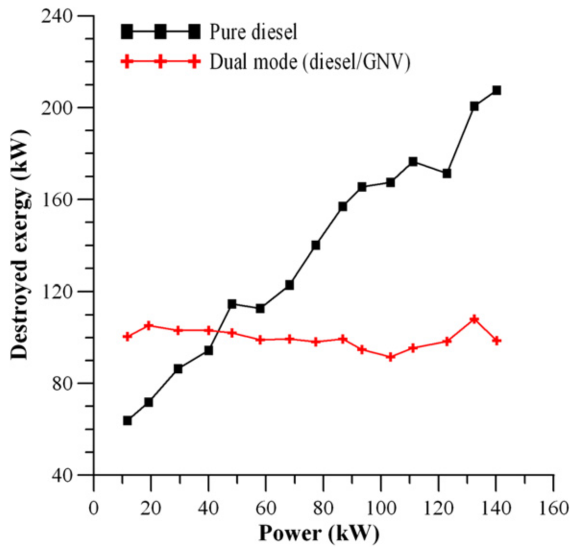

Fig. 9. Destroyed exergy for pure diesel and dual mode as a function of the power.

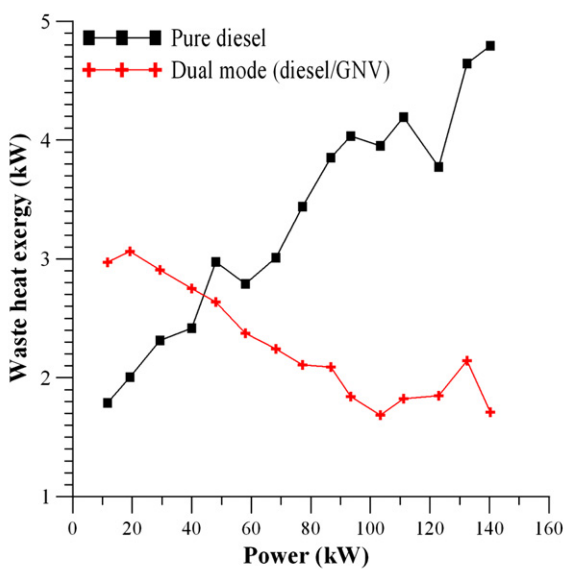

Fig. 10. Waste heat exergy for the environment as a function of the power.

combustion process is highly irreversible. The results are not essentially different of the energetic efficiency, because the exergy of fuel differs only slightly from their combustion enthalpy.

Combustion is a process that converts chemical energy of fuel and oxygen into the sensible enthalpy of combustion products, generating entropy and consuming energy. This process usually occurs simultaneously with heat transfer as well as fluid friction and mixing. Analysis reported in this work includes both chemical reaction and the heat transfer. Irreversibility's due to friction and mixing are negligible.

The total exergy for the engine operating in the diesel and dual condition is presented in Fig. 8. It is verified that total exergy for the pure diesel condition is higher than that of dual mode for power 50 kW, reaching value of 425.52 kW for the power 150 kW. In the dual mode, total exergy is lesser than that of pure diesel for powers up to 50 kW and growing until the value of 287.35 kW for the power 150 kW.

Fig. 9 shows the behavior of destroyed exergy for the diesel and dual mode condition. For the case where we use pure diesel, it was observed that there is a large destruction of exergy, reaching up to values of 207 kW to 150 kW power. For the dual condition it is verified that for the power of 10 kW, the values are 100 kW and growing until the value of 106.2 kW for the power of the engine 150 kW. The heat exergy transferred to the environment as a function of power is shown in Fig. 10, where we can see that pure diesel exergy reach 4.7 kW to power of 150 kW and 1.7 kW for engine power of 10 kW. For dual-mode operation there is a decrease of exergy from approximately 3 kW to 10 kW of engine power to approximately 1.7 kW to 150 kW of engine power. These results show that the heat exergy represents a small part of total exergy ($\approx 2\%$).

Fig. 11 shows the behavior of the exergy of the combustion products to pure diesel and dual mode. We can check that there is growing of exergy for all conditions of operation (diesel and dual mode). For the pure diesel, exergy change from 7.47 kW in the power of 10 kW to 62.58 kW in the power of 150 kW. For operation in dual-mode, we can see that exergy starting with values of 15.12 kW for the engine power of 10 kW, reaching values of 39.41 kW to 90 kW and decreasing for values of 36.48 kW at the engine power of 150 kW. It is important to note that for engine power up 90 kW, there is now a considerable exergetic potential, which can provide a condition of use of exergy in co-generation system and heat and cold production.

In order to improve understanding, the energy and exergy balance can be shown in a flow diagram. Figs. 12–16 illustrate the representation proposed in this work. Figs. 12 and 13 show the flow exergy predicted by the model for the operating conditions

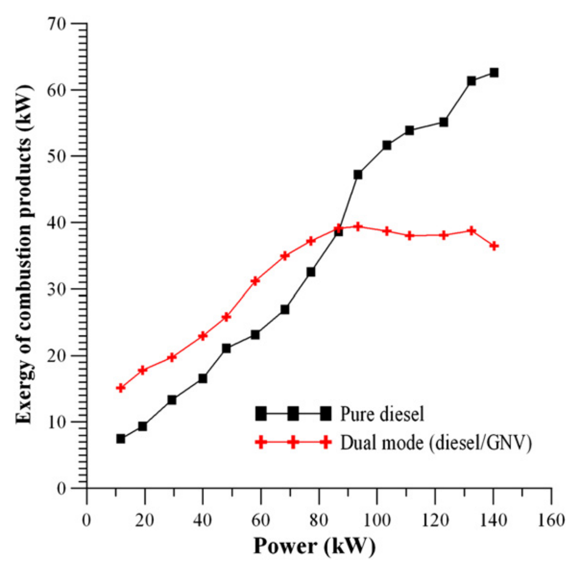

Fig. 11. Exergy of the combustion products as a function of the power.

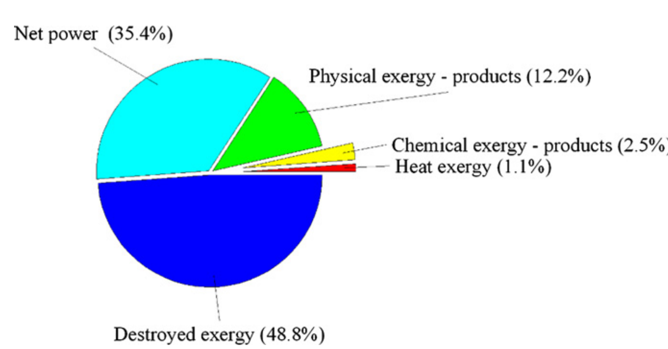

Fig. 12. Diagram of flow exergy for pure diesel condition at the higher load total (total exergy 425.52 kW).

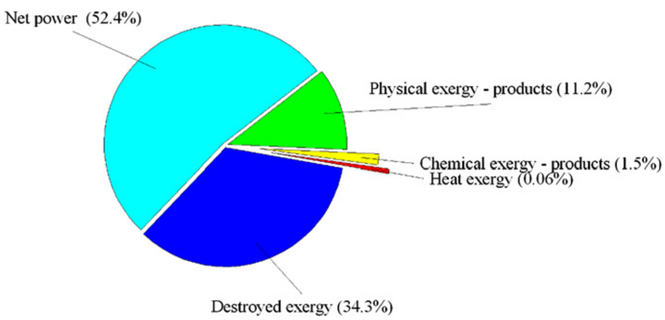

Fig. 13. Diagram the flow exergy with dual condition at the higher load (total exergy 287.35 kW).

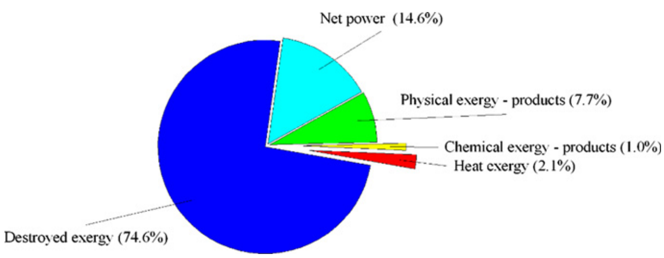

Fig. 14. Diagram of flow exergy for pure diesel at condition the lower load (total exergy 85.68 kW).

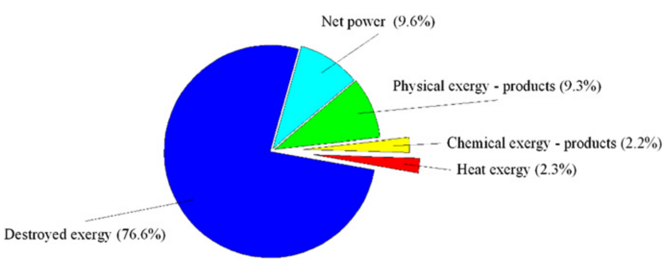

Fig. 15. Diagram of flow exergy for dual condition at the lower load (total exergy 131.07 kW).

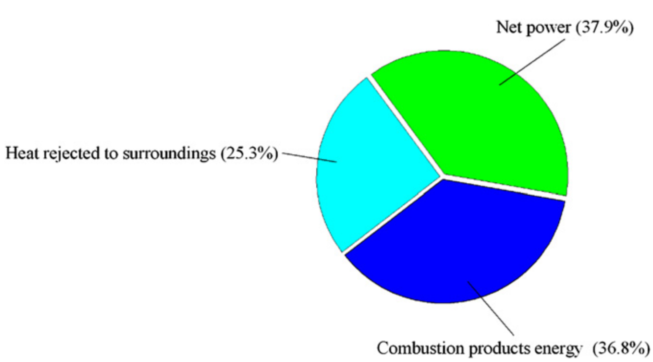

Fig. 16. Diagram of flow energy for pure diesel condition at higher power (total energy 396.78 kW).

with pure diesel and dual fuel for higher power. While the Figs. 14 and 15 present the same results for lower power conditions. Exergetic analysis shows that the main sources of exergy losses are irreversibility of combustion and exhaust gas emissions of the engine. In the pure diesel mode, because of the higher exhaust temperature, the exergy loss is higher than in dual mode operation. Heat exergy losses are relatively small ($<2.5\%$ of input exergy).

Figs. 16 and 17 show the flow energy diagram for the operating conditions of higher power, using pure diesel and dual

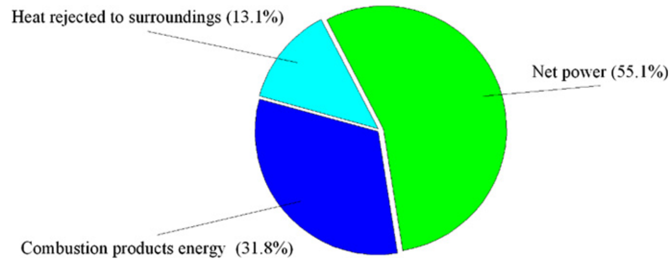

Fig. 17. Diagram of flow energy for dual condition at higher power (total energy 273.1 kW).

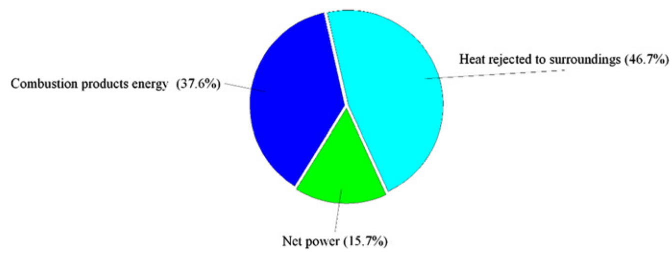

Fig. 18. Diagram of flow energy for pure diesel condition at lower power (total energy 79.89 kW).

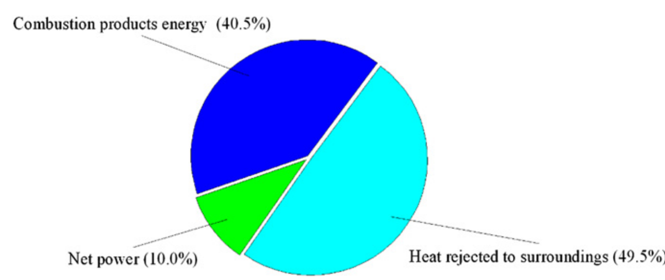

Fig. 19. Diagram of flow energy for dual condition at lower power (total energy 125.26 kW).

fuel, respectively. It is observed that for pure diesel condition, (396.78 kW fuel energy), we obtain values of the performance that agree well with literature [21], which refer to values close to 1/3 of waste heat, 1/3 to produced power and $^1/_3$ to products energy. Pereira et al. [42] presents values of 41.5% net power, 36.1% exhaust gas energy and 27% to other losses.

For dual mode, 273.01 kW fuel energy, heat and combustion products energy losses are lower than pure diesel condition. This becomes the use of the engine in the dual mode, viable economically.

Figs. 18 and 19 present a diagram of energy for the pure diesel and dual-mode conditions at lower power. By using diesel pure at lower power we see that useful power, heat loss for the environment and products energy are considerably lower than the higher power condition.

Exergy is not conserved, and the irreversibility's of the engine, such as combustion, heat transfer, mixing, friction, etc, destroy a significant fraction of the fuel exergy. The relative external loss of the thermal exergy (physical exergy plus chemical exergy) of the combustion gases is smaller than the relative loss of combustion gases enthalpy. According Szargut [28], the comparison between energy and exergy losses in the engine leads to the following general comments:

(a) The main causes that contribute to the imperfection in the mechanical power production cannot be detected by means of an energetic analyze;
(b) The main exergy losses have no equivalent in the energetic analyses;

(c) The losses of energy appearing in the energetic analyze (heat loss and exhaust gas enthalpy) have relatively smaller values in the exergy analyzed;
(d) In the exergy analyze heat absorption by the surrounding is not accompanied by an exergy loss; it is the temperature difference between the boundary of heat rejected and the surrounding that determines the magnitude of the exergy loss. Since the exergy of the environment can never increase, heat transfer process leads only to dissipation of exergy;
(e) The exergy losses in the engine can be reduced by raising the combustion gas temperature and by using the combustion gases or lost heat in another processes such as pre heating combustion reactants, chemical recuperation, production of steam or hot water for external consumes, and refrigeration.

5. Conclusions

In this paper, theoretical and experimental studies of energy and exergy have been explored in dual-fuel engine (natural gas and diesel). Interest in this type of physical problem is motivated by its importance in many practical situations, especially to electrical energy generation. From the results, the following conclusion can be cited:

(a) Viability of use of diesel engine to operate in dual mode with natural gas was verified.
(b) The engine was operated of satisfactory form and it had been reached replacement rate more than 80% without by presenting any abnormality such as detonation phenomenon.
(c) The mathematical model of energy and exergy predict satisfactory the process as compared with experimental values and literature data.
(d) The energy efficiency ranged from 15.7 to 37.9% in diesel pure mode and 10.02 to 55.13% in dual mode, to power changing from 10 to 150 kW.
(e) The exergetic efficiency ranged from 14.6 to 35.4% in diesel pure mode and from 9.57 to 52.38% in dual mode, when the engine power changes from 10 kW to 150 kW.
(f) For the engine operating with pure diesel, total exergy presented values ranging from 85 kW to 425 kW, for engine power ranging from 10 to 150 kW. When it has operated in dual mode, the total exergy presented values varying from 131.0 to 287.3 kW.
(g) The destroyed exergy ranged from 100.4 to 98.6 kW, the exhaust gas exergy ranged from 15.1 to 36.4 kW, while the waste heat to environment presents values varying from -2.9 to -1.7 kW, respectively for power changing from 10 to 150 kW, when the engine operated in dual mode.
(h) The destroyed exergy ranged from 62 to 206 kW, the exhaust gas exergy ranged from 8 to 62 kW, while the waste heat to environment presents values varying from -1.7 to -4.8 kW for power changing from 10 to 150 kW, when operated in pure diesel.

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